RAY

Design of a Five Ton Electric Pillar Crane

Mechanical Engineering

B. S.

1915

DESIGN OF A FIVE TON ELECTRIC PILLAR CRANE

BY

HUGH LIGHT RAY

THESIS

FOR THE

DEGREE OF BACHELOR OF SCIENCE

IN

MECHANICAL ENGINEERING

IN

THE COLLEGE OF ENGINEERING

OF THE

UNIVERSITY OF ILLINOIS

1915

Certificate

TABLE OF CONTENTS.

Chapter 2. Calculations.

Plates.

DESIGN

OF A

5 TON ELECTRIC PILLAR CRANE.

Introduction.

The object of this thesis is to design in its entirety a 5 Ton Electric Pillar Crane in accordance with the specifications as stated on page 2. The crane throughout is to be designed in a high grade manner, all details being in harmony with the use of cut gearing. Among the references used in this design are notes on machine design and mechanics of machinery used in the Mechanical Engineering Department of the University of Illinois, Machine Design by Hess, Kent's Hand Book and Cambria Hand Book. Numerous catalogues, photographs, specifications and drawings of several crane builders illustrating American practice have been examined. The stresses in the frames and machinery have been determined by the principles of mechanics and the design carried out on a theoretical rather than an empirical basis.

Chapter I.

Specifications.

I.-Power.- Electric motor for hoisting only. Hand power for rotating.

2.-Capacity.- Maximum hoisting load 5 tons.

3.-Dimensions.- Effective radius I6'-0". Lift I5'-0".

4.-Material.- The frame and miscellaneous structural material will be made of medium steel to conform to the specifications adopted by the Association of American Manufacturers. All iron castings will be of tough material, free from injurious cold ducts or blow holes, true to pattern and of a workmanlike finish.

5.-Factor of Safety.- This crane to be proportioned so that the factor of safety will in no case be less than five (5) when the crane is operated to its full capacity. This factor is based on the ultimate strengh of material used. For the cast iron pillar the stress in tension should not exceed 3000# per sq.in.

6.-Hoisting.- The hoisting gearing to be arranged for operation with two speeds. A slow speed for heavy loads and a fast speed for light loads.

7.-Gearing.- Gearing throughout to be extra strong, wide face spur gears. All gears except the drum gear and its mating pinion being cut from solid stock. Gear bearings throughout to be bronze bushed and provided with suitable provisions for lubrication.

8.-Drum.- The drum to be machine turned and grooved and of ample size to take without overlapping the effective length of the hoisting chain.

9.-Brakes.- Crane to be equipped with an automatic safety brake so designed, that the load may be held at any point, that the change of speed may be made while the load is being hoisted and that the load in lowering can not run away from the motor. In addition a dispatch lowering brake controlled by a hand wheel enabling the operator to lower the load independent of the motor.

10.-Block and Sheaves.- Block to be of neat design having a heavy forged hook with head designed to turn on bronze washer. Sheaves to be of large diameter having turned grooves to insure smooth running of hoisting chain and to be bushed with bronze.

II.-Frame and Pillar.- Jib and platform for supporting motor to be constructed of structural steel shapes and properly fastened together. Frame to be supported by a heavy cast iron pillar having a base of large diameter and provided with bolts for fastening the pillar to the foundation. The thrust of the frame to be carried by rollers,bearing against a turned roller path on the pillar The revolving top cap of pillar to be of cast iron bearing on an anti-friction washer. Top cap to be secured to platform and outer end of jib by means of suitable steel tie rods.

I2.-General.- The crane throughout is to be designed in a high grade manner, all details being in harmony with the use of cut gearing.

CHAPTER II.

Calculations.

I3.-Proportions.- An inspection of a large number of pillar cranes from catalogues showed that the prevailing angle of the jib was 45°. The upper tie rods approximated an angle of 20° with horizontal. Fig. I, page 6 gives the fixed dimensions and lines of action assumed in this design.

I4.-Chain.- Fig. 6, page 8 shows the positions of chain pulls S_1, S_2 and S_3. For floating block supporting load $S_2 = I.03\ S_1$. $S_1 + S_2 = I0,000\#$. $S_1 = 4,925\#$. $S_2 = 5,075\#$. For fixed block at end of boom $S_3 = I.05 S_2$. $S_3 = 5,350\#$. This is the load which the chain must support. According to Hess, Machine Design- Page 63, the ordinary safe load for general use of a $\frac{9''}{16}$ chain is $5,600\#$. Size of chain $\frac{9''}{16}$. Outside width 2".

I5.-Drum.- Size of chain $\frac{9''}{16} = d$. For power driven drums the diameter (D) should equal $25 \times d = I4.I''$, call it I5".
Width of grooves $= d + \frac{1''}{16} = \frac{5''}{8}$. The drum thickness under the grove is approximately $\frac{D}{18} + \frac{3''}{8} = I.20''$, call it I". Depth of grooves $b = I\frac{1}{4}d + \frac{1''}{16} = \frac{49''}{64}$, call it $\frac{3''}{4}$. Pitch of grooves $= 3\frac{1}{2}d + \frac{1''}{8} = 2\frac{3''}{32}$.
Maximum rise of hook I5'-0". Length of chain to be rolled on drum $= 2 \times I5 = 30'\text{-}0''$. Circumference of one turn $= \frac{\pi \times I5}{I2} = 3.93$ ft. Number of

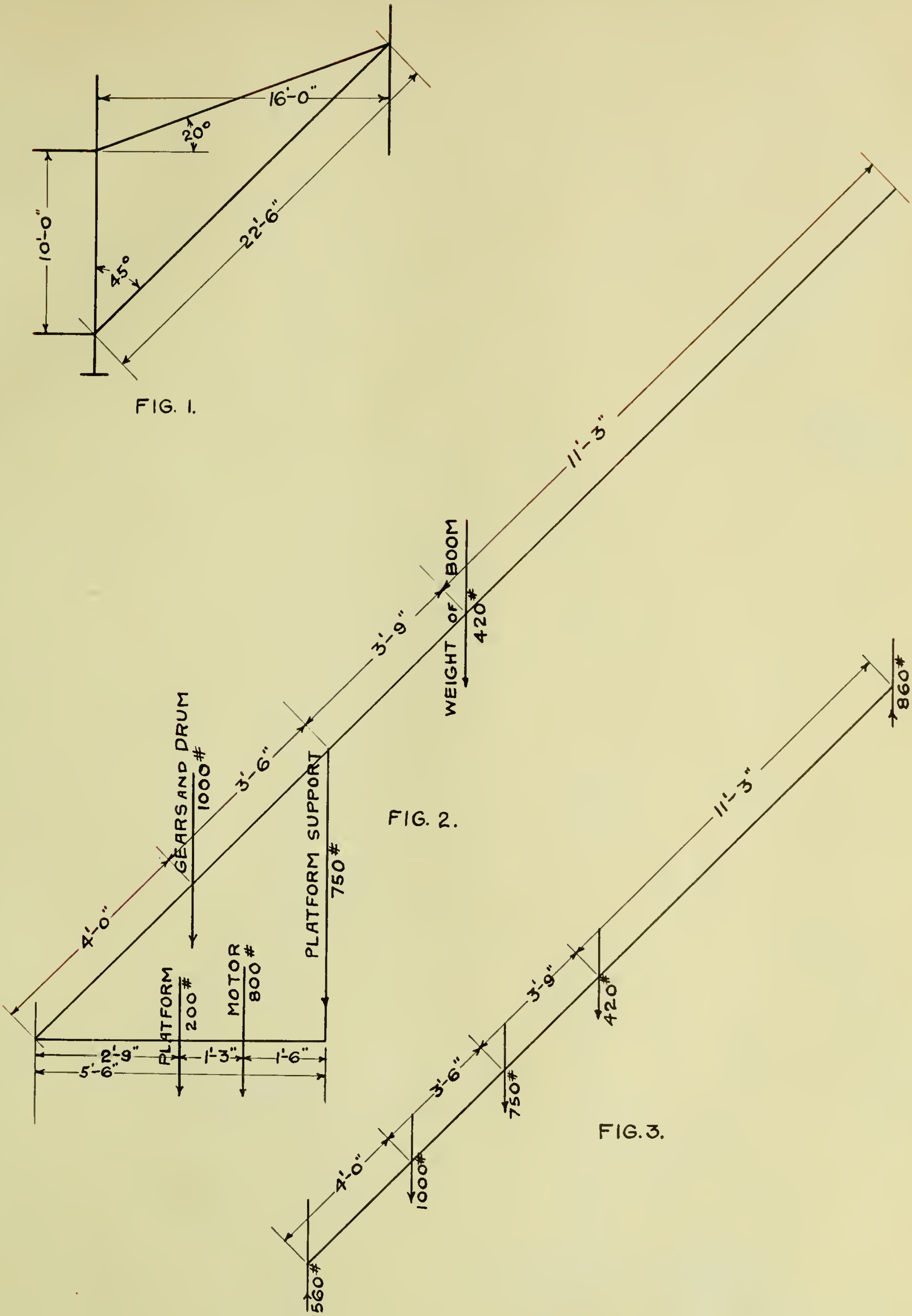


FIG. 1.

FIG. 2.

FIG. 3.

PLATE I.

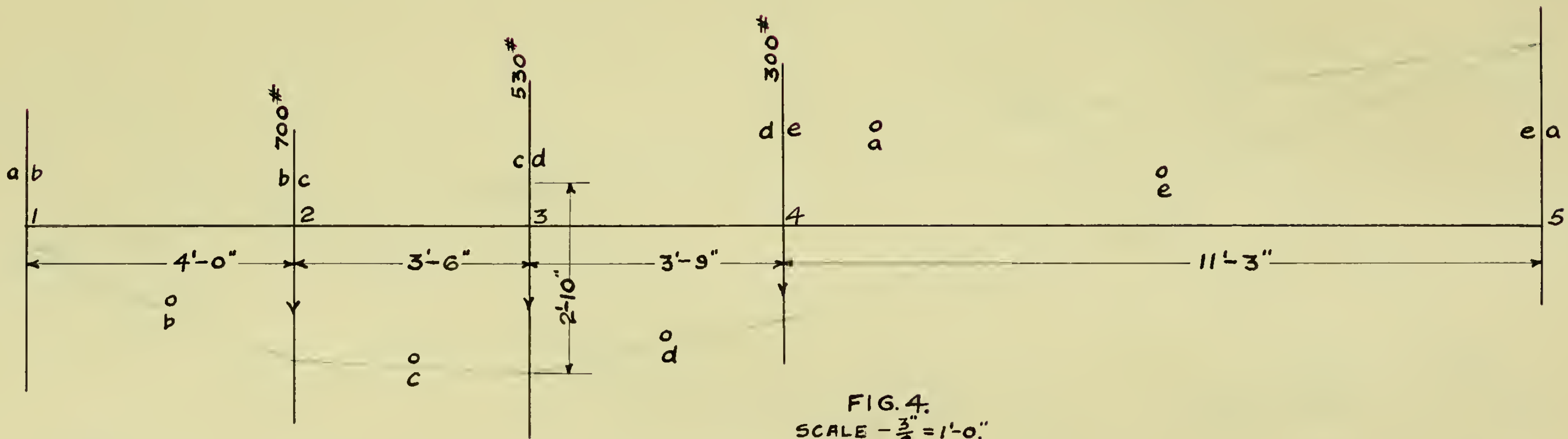


FIG. 4.
SCALE - $\frac{3}{8}$" = 1'-0."

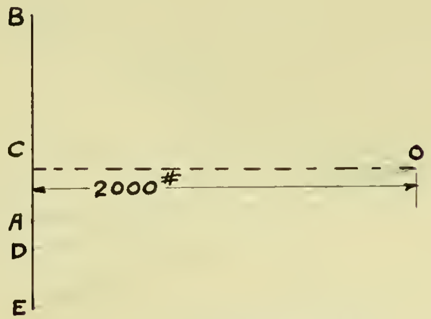


FIG. 5
SCALE - 1" = 1000#

PLATE 2.

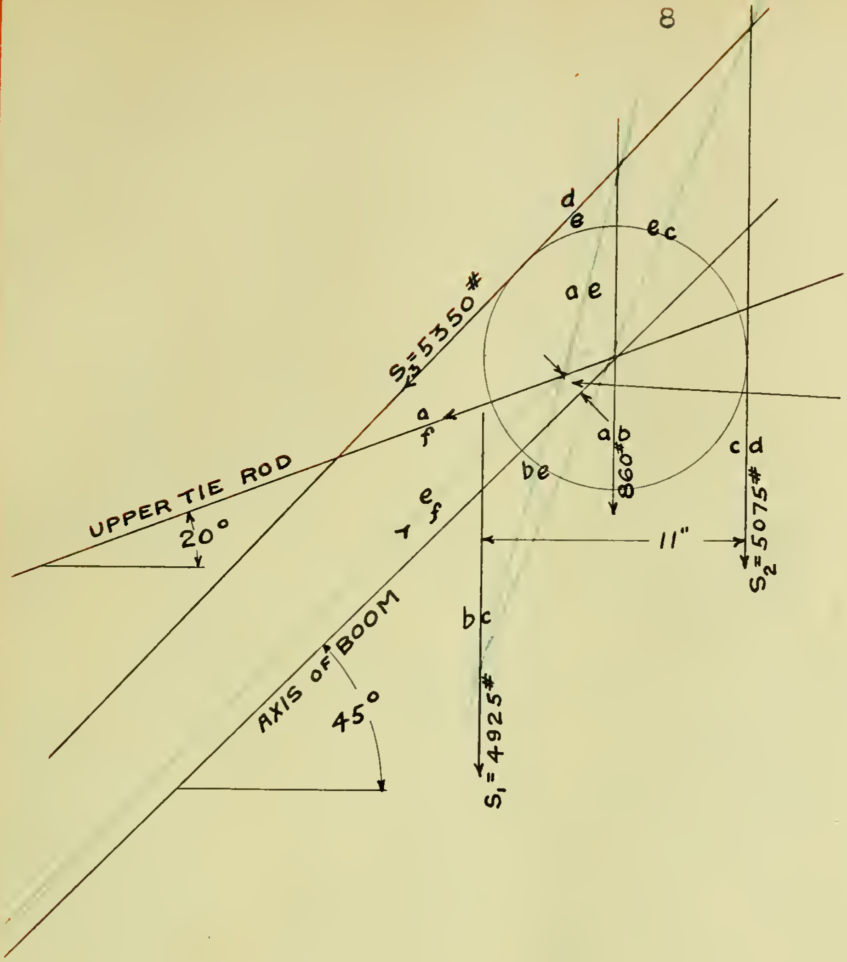


FIG. 6.
SCALE – $1\frac{1}{2}$" = 1'-0".

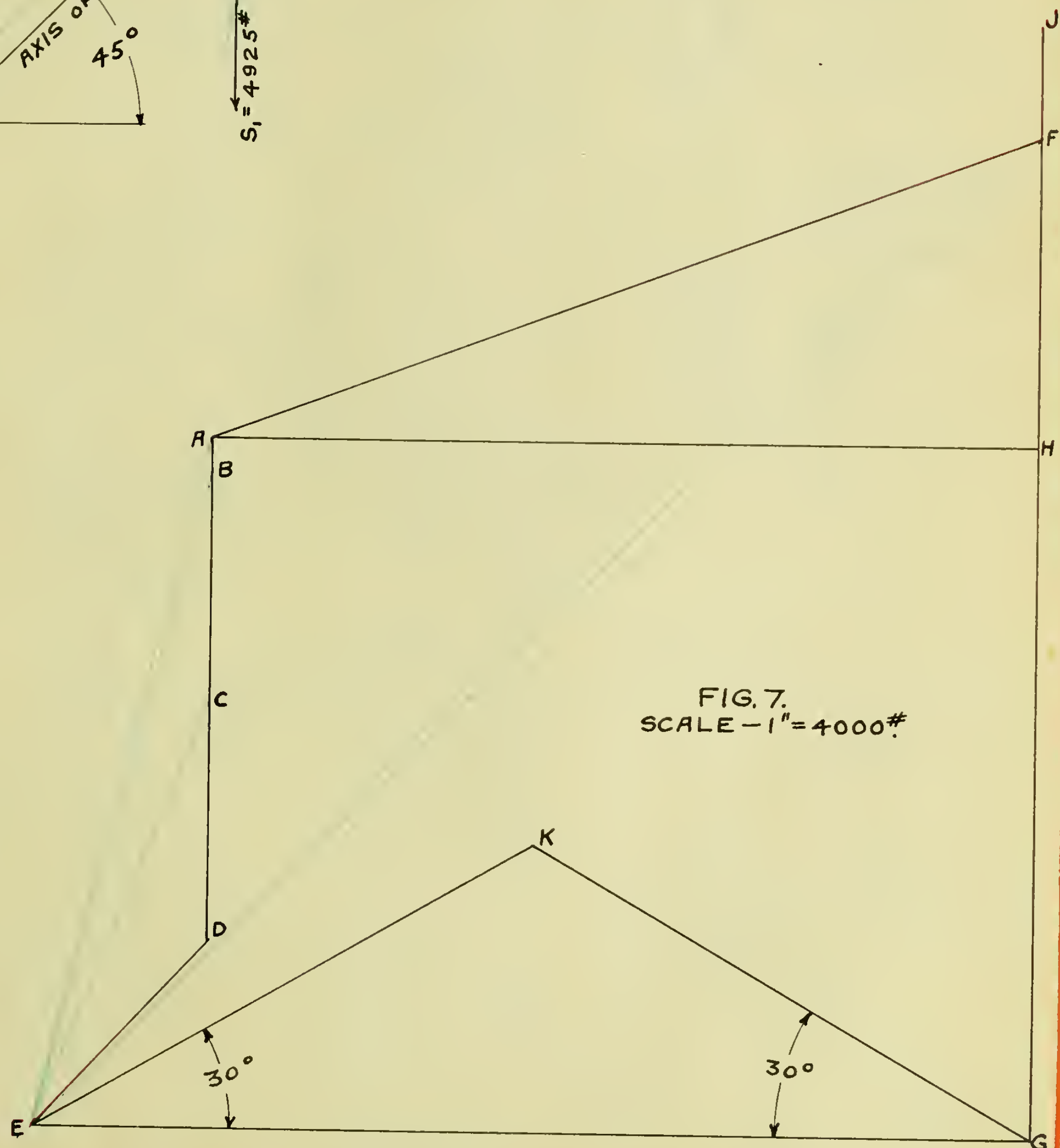
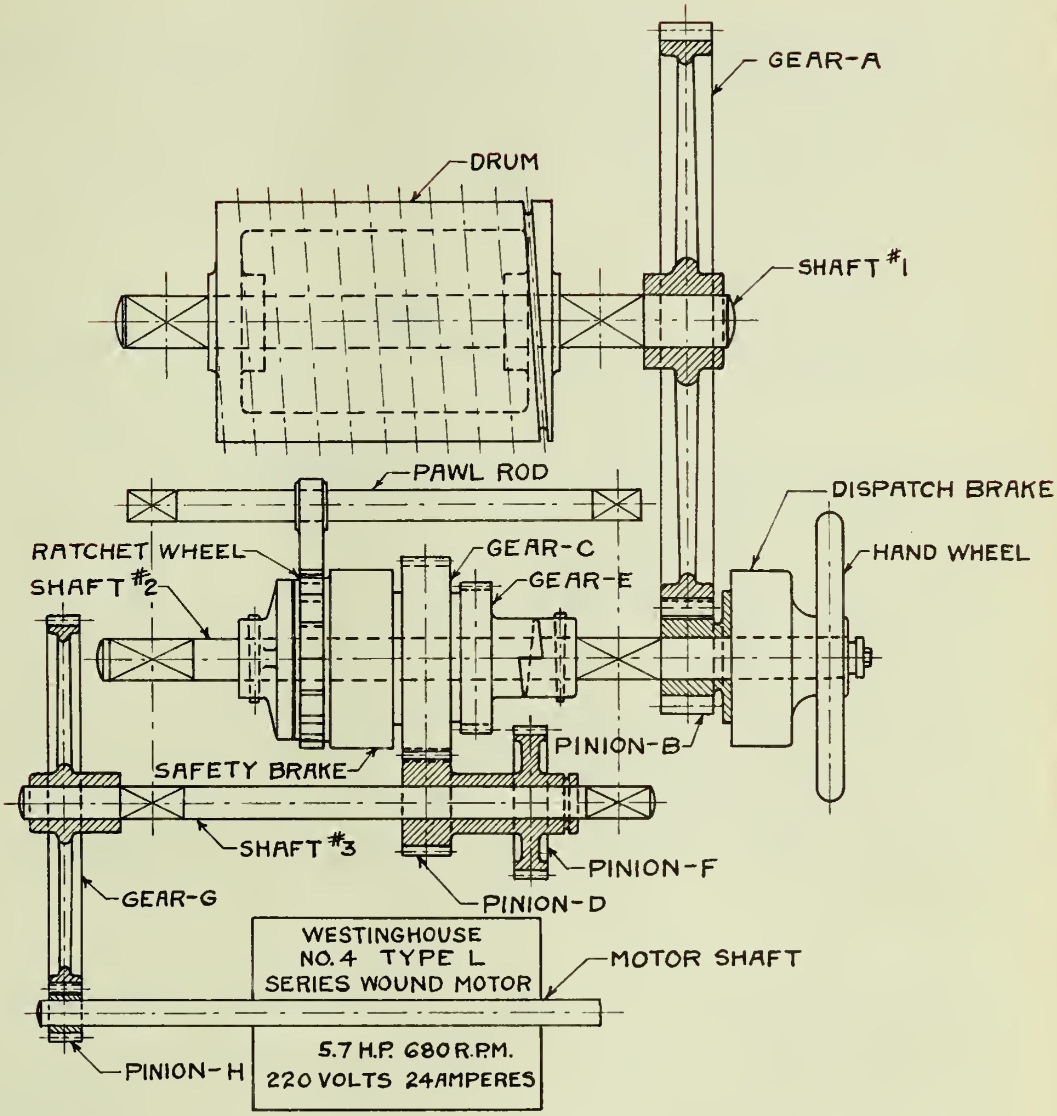


FIG. 7.
SCALE – 1" = 4000#.

PLATE 3.



SCALE 1" = 1'-0"

PLATE 4

turns required is $\frac{30}{3.93}=7.65$. Allowing one turn for fastening and one turn for over-winding, the total number of turns required will be $7.65+2=9.65$, say IO turns. Width of drum is $10\times2\frac{3''}{32}=20.93''$, call it 2I".

I6.-Sheave Wheels.- The diameter of the wheel will be $20\times d=11.2''$, call it II". Depth of groove same as for drum equals $\frac{3''}{4}$. Width of groove same as for drum equals $\frac{5''}{8}$.

I7.-Motor.- The efficiency of the hoisting apparatus is assumed as 80%. Speed of hoisting I5 ft. per min.

H.P. of the motor will be $\frac{10,000\times5}{33,000\times0.80}=5.7$. A Westinghouse, No 4, Type L, series wound motor using 220 volts was selected. When delivering 5.7 H.P. the R.P.M. will be 680 and current used is 24 amperes. Velocity of rope running on drum is $15\times2=30$ ft. per min. R.P.M. of drum is $\frac{30\times12}{\pi\times15}=7.65$. The reduction from the motor to the drum is $\frac{680}{7.65}=89$, call it 90. This is obtained by three reductions of 7.5:I, 2:I and 6:I respectively. A diagram of the winch is shown on page 9.

I8.-Drum Gear and Pinion.- These are designated as Gear A and Pinion B on page 9. Twisting moment of shaft #2 carrying pinion B is $\frac{5400\times7.5}{6\times0.97\times0.97\times0.96}=7,350$ in. lbs. Assuming I2 teeth for pinion B the circular pitch is given, when the width of face is $2p_c$, by

$$p_c=\sqrt[3]{\frac{6.28\times7,350}{12\times7,150\times2\left(0.124-\frac{0.684}{12}\right)}}=1.57'', \text{ say 2 pitch.}$$

Gear A, 36" p.d.; 2 pitch; 72 teeth; $3\frac{1}{4}''$ face; cast iron.

Pinion B, 6" p.d.; 2 pitch; I2 teeth; $3\frac{1}{4}''$ face; cast iron.

Force acting at pitch line is $\frac{7,350}{3} = 2,400\#$. Assuming 6 gear arms of elliptical section and S as 5,000# the width of arm at hub will be

$$h = \sqrt[3]{\frac{20 \times 2,400 \times 18}{5,000 \times 6}} = 3.07", \text{ call it } 3".$$

Gear A will have 6 arms, $3" \times 1\frac{1}{2}"$ at the hub and $2" \times 1"$ at the rim.

19.-Intermediate Gear and Pinion.- These are designated as Gear C and Pinion D on page 9. Twisting moment of shaft #3 carrying pinion D is $\frac{7,350}{2 \times 0.98 \times 0.97} = 3,780$ in. lbs. Assuming 24 teeth and width of face $4p_c$ for pinion D we have

$$p_c = \sqrt[3]{\frac{6.28 \times 3,780}{24 \times 5,500 \times 4\left(0.124 - \frac{0.684}{24}\right)}} = 0.78", \text{ call it 4 pitch.}$$

Gear C, 12" p.d.; 4 pitch; 48 teeth; 3" face; cast iron.

Pinion D, 6" p.d.; 4 pitch; 24 teeth; 3" face; cast iron.

20.-Auxilliary Gear and Pinion.- These are designated as Gear E and Pinion F on page 9. The speed reduction from shaft #2 to shaft #3 instead of being 2:1, will be 1;1. The twisting moment of shaft #3 will be the same as for full load for while the load is one half full load the reduction has been cut down in exact proportion to the change in load. The force acting at the pitch line will be $\frac{24}{36}$ of that for Gear C and Pinion D, if the same diametral pitch is used.

Gear E, 9" p.d.; 4 pitch; 36 teeth; 2" face; cast iron.

Pinion F, 9" p.d.; 4 pitch; 36 teeth; 2" face; cast iron.

21.-Motor Gear and Pinion.- These are designated as Gear G and Pinion H on page 9. Twisting moment of motor shaft is $\frac{3,780}{7.5 \times 0.98 \times 0.97} = 530$ in. lbs. Assuming 15 teeth and width of face

$3p_c$ for pinion H, we have

$$p_c = \sqrt[3]{\frac{6.28 \times 530}{15 \times 4{,}000 \times 3\left(0.124 - \frac{0.684}{15}\right)}} = 0.6", \text{ use 5 pitch.}$$

Gear G, 23" p.d.; 5 pitch; 115 teeth; 2" face; cast iron.

Pinion H, 3" p.d.; 5 pitch; 15 teeth; 2" face; cast iron.

Bore of hole in pinion H, for motor shaft $1\frac{5}{8}$". Size of key $\frac{3}{8}" \times \frac{1}{4}"$.

Force acting at pitch line is $\frac{530}{1.5} = 350\#$. Assuming 6 gear arms of elliptical section and S as 3,000#, we have

$$h = \sqrt[3]{\frac{20 \times 350 \times 11.25}{3{,}000 \times 6}} = 1.63", \text{ say } 1\tfrac{3}{4}".$$

Gear G will have 6 arms $1\frac{3}{4}" \times \frac{7}{8}"$ at the hub and $1\frac{1}{4}" \times \frac{5}{8}"$ at the rim.

22.-Drum Shaft.- This is shown as shaft #1 on page 9. Twisting moment of shaft is $\frac{5{,}400 \times 7.5}{0.97 \times 0.98} = 42{,}500$ in. lbs.

Allow S as 12,000#. $\frac{42{,}500}{12{,}000} = 3.55$. From tables d for twisting should be 2.625". Assume length of bearing as 5". Distance between centers of bearings is $5 + 22 = 27"$. Bending moment of shaft at center of drum will be $\frac{5{,}400 \times 13.5}{2} = 35{,}000$ in. lbs. $\frac{M}{T} = \frac{35{,}000}{42{,}500} = 0.82$. From tables n was found to be 1.29. Diameter of shaft for combined bending and twisting is $1.29 \times 2.625 = 3.4"$, call it $3\frac{1}{4}"$. Thickness of bushing $\frac{5}{16}"$.

23.-Intermediate or Brake Shaft.- This is designated as shaft #2 on page 9. The twisting moment was found to be 7,350 in. lbs. $\frac{7{,}350}{12{,}000} = 0.61$. From tables d for twisting should be 1.5". Force acting at pitch line of gear C and pinion D is $\frac{3{,}780}{3} = 1{,}260\#$.

Bending moment on shaft at gear C will be $\frac{1{,}260 \times 1.5 \times 12}{2.5} = 9{,}000$ in. lbs.

Force acting at pitch line of gear A and pinion B is 2,400#.

Bending moment of shaft at bearing, due to pinion B, will be found by

$2{,}400 \times 4 = 9{,}600$ in. lbs. $\frac{M}{T} = \frac{9{,}600}{7{,}350} = I.3$. From tables n is I.4. Diameter of shaft for combined bending and twisting will be $I.5 \times I.4 = 2.I''$. As this shaft will be weakened somewhat by holes necessary for safety brake make this deameter $2\frac{1}{2}''$. Thickness of bronze bushing $\frac{5}{16}''$.

24.-Speed Gear Shaft.- This is shown as shaft #3 on page 9 The twisting moment was found to be 3,780 in. lbs. $\frac{3{,}780}{12{,}000} = 0.3I5$. From tables d is $I\frac{3}{16}''$. Bending moment is the same as for shaft #2 at gear C and is 9,000 in. lbs. $\frac{M}{T} = \frac{9{,}000}{3{,}780} = 2.4$. From tables n is I.7I. Diameter of shaft for combined twisting and bending will be $I.7I \times I.I875 = 2.03$, call it 2". Thickness of bronze bushing $\frac{1}{4}''$.

25.-Jib.- Fig. 2, page 6 shows the dead loads acting on the frame and Fig. 3 the loads on the jib. Fig. 4, page 7 shows that the maximum bending moment due to these loads takes place at point 3 and is 68,000 in. lbs. Fig. 6, page 8 shows the various forces acting at the head of the jib. The resultant of these forces has an eccentricity of I" with the axis of the jib. Fig. 7, page 8 is the force polygon for the forces acting on the frame. Stress in jib is 30,500#. Bending moment at point 3, Fig. 4, due to eccentricity will be $\frac{30{,}500 \times 7.5}{22.5} = I0{,}000$ in. lbs. Total bending moment at point 3 is $I0{,}000 + 68{,}000 = 78{,}000$ in. lbs. Bending moment in one channel will be $\frac{78{,}000}{2} = 39{,}000$ in. lbs. Direct load in one channel is $\frac{30{,}500}{2} = I5{,}000\#$.

Assume $2 - 8'' - 2I.25\#$ channels. $r = 2.77$. l is $I2 \times 22.5 = 270''$. Area of section 6.25 sq. in. Allowable stress as a column is

$$\frac{I4{,}000}{I + \frac{(270)^2}{I0{,}000 \times (2.77)^2}} = 7{,}200\#$$

Actual stress is

$$\frac{15{,}000}{6.25}+\frac{39{,}000\times 4}{47.8-\dfrac{15{,}000\times(270)^2}{10\times 28{,}000{,}000}}=6{,}000\#.$$

This value is low but the channels will be weakened somewhat by holes made to fasten machinery. r about minor axis is 2.77. $\frac{l}{r}=\frac{270}{2.77}=97.5$. To prevent these struts from failing about the minor axis the two channels must be fastened together at intervals of not to exceed $97.5\times 0.6=58.5''$, say 4'-10".

26.-Pillar.- As shown by Fig. 7, page 8 the horizontal component of the forces in the top tie rods is $AH=18{,}000\#$. The diameter of the pillar at the upper bearing will be determined by the size of the pin. Length of bearing assumed as 6". Thickness of bronze washer I". Bending moment on pin $=(3+1)\times 18{,}000=72{,}000$ in. lbs Allowable stress 15,000#. $\frac{72{,}000}{15{,}000}=4.8$. From tables d is $3\frac{5}{8}''$. Pressure per sq. in. of projected area of bearing will be $\frac{18{,}000}{6\times 3.62}=830\#$. This is high and the pin will be made 5" in diameter in the pillar and 6" in the pillar head. Pressure per sq. in. is $\frac{18{,}000}{6\times 6}=500\#$, which is satisfactory for intermittant and reverse pressures. Bending moment on a section I2" below the center of bearing is $18{,}000\times 12=216{,}000$ in. lbs. Allowable stress in tension for cast iron 3,000#. Direct load 15,000#. $S_2=\frac{15{,}000}{75}=200\#$. Outside diameter 12".
$216{,}000=S_1\times 0.1\left(\frac{(12)^4-(7)^4}{12}\right)$. $S_1=1{,}420\#$. $S=S_1+S_2=1{,}620\#$.
A uniform thickness of metal of $2\frac{1}{2}''$ for a hollow round cylindrical section is assumed. Bending moment on section at middle of pillar is $5\times 12\times 18{,}000=1{,}080{,}000$ in. lbs. Outside diameter I8". Direct load 15,000#. $S_2=\frac{15{,}000}{122}=125\#$. $1{,}080{,}000=S_1\times 0.1\left(\frac{(18)^4-(13)^4}{18}\right)$. $S_1=2{,}520\#$.

$S = S_1 + S_2 = 2{,}645\#$.

Bending moment on a section at the foot of pillar is $10 \times 12 \times 18{,}000 =$ $2{,}160{,}000$ in. lbs. Outside diameter 24". Direct load $15{,}000\#$.

$S_2 = \frac{15{,}000}{170} = 90\#$. $2{,}160{,}000 = S_1 \times 0.1\left(\frac{(24)^4 - (19)^4}{24}\right)$. $S_1 = 2{,}600\#$.

$S = S_1 + S_2 = 2{,}690\#$.

Weight of pillar figured to be $5{,}000\#$.

Weight of boom figured to be $500\#$.

Weight of gear and drum figured to be $1{,}000\#$.

Bolt reaction is $\frac{(14 \times 10{,}000) + (6 \times 500) + (1 \times 1{,}000) - (5{,}000 \times 2)}{4} = 34{,}500\#$.

This is on the theory that the entire load due to overturning is taken by one bolt. It will be assumed in this design that of the 8 bolts acting each one will support in proportion to its distance at right angles from a straight line passing through the point of overturning. This line in turn being at right angles to the axis of the boom. In this case the bolt farthest from this point would support $\frac{1}{4}$ of the load or $\frac{34{,}500}{4} = 9{,}000\#$. Allowable fibre stress $12{,}000\#$. Area at the root of thread is $\frac{9{,}000}{12{,}000} = 0.75$ sq. in., say $1\frac{1}{4}''$ bolts. Bending moment on the arm section of base near the hub is $9{,}000 \times 10 = 90{,}000$ in. lbs. Allowable fibre stress $3{,}500\#$.

$\frac{I}{C} = \frac{90{,}000}{3{,}500} = 25$. $\frac{I}{C}$ of section assumed $\left(\frac{4 \times (4.5)^3}{12} + \frac{7 \times (2.5)^3}{12}\right) \times \frac{1}{1.5} = 24$.

27.-Top Cap.- Force acting in side tie rods is shown in Fig. 7, page 8 to be $GJ = 24{,}000\#$. tension. Force in one rod is $\frac{24{,}000}{2} =$ $12{,}000\#$. Bending moment on arm of pillar head is $12{,}000 \times 5 = 60{,}000$ in. lbs. Allowable stress taken as $3{,}500\#$. $\frac{I}{C} = \frac{60{,}000}{3{,}500} = 17$. Assume width of section as 4". $17 = \frac{bd^2}{6}$. $d = 5.05''$, say 5".

28.-Top Cap Bearing.- Area of bronze washer at top bearing will be $\frac{\pi((10)^2-(6)^2)}{4}=50$ sq. in. Load is 15,000#. Pressure per sq. in. of area of bearing is $\frac{15{,}000}{50}=300\#$. This is satisfactory.

29.-Top Tie Rods.- Force acting in top tie rods is shown in Fig. 7, page 8 to be AF = 19,000#, tension. Force in one rod is $\frac{19{,}000}{2}=9{,}500\#$. Allowable fibre stress taken as 12,000#. Area at root of threads will be $\frac{9{,}500}{12{,}000}=0.81$ sq. in. This will require rods 1" in diameter. Diameter of upset screw ends $1\frac{3}{8}$".

30.-Side Tie Rods.- Force acting in the side tie rods was found to be 24,000#. Force in one rod is $\frac{24{,}000}{2}=12{,}000\#$. Allowable fibre stress taken as 12,000#. Area at root of threads will be $\frac{12{,}000}{12{,}000}=1.0$ sq. in. This will require rods $1\frac{1}{8}$" in diameter. Diameter of upset screw end $1\frac{1}{2}$".

31.-Rollers.- Normal pressure between roller and post shown in Fig. 7, page 8 to be EK = 12,500#. Diameter of rollers 8", length $7\frac{3}{4}$". Bending moment on pin will be $\frac{12{,}500\times 8.5}{2\times 2}=26{,}600$ in. lbs. Allowable fibre stress 12000#. $\frac{26{,}600}{12{,}000}=2.21$. From tables d is 2.81", call it 3". Pressure per sq. in. of projected area of pin in roller $\frac{12{,}500}{7.75\times 3}=540\#$, this is satisfactory. Allowable pressure on roller for cast iron roller is $350\times 5\times 8=14{,}000\#$. (Bethman)

32.-Hook.- The hook was designed according to dimensions given by Unwin. Load is 10,000#. Area at root of thread will be $\frac{10{,}000}{8{,}000}$

$= 1.25$ sq. in. This would require $1\frac{1}{4}''$ bolts. Diameter of material from which hooks is made is given, when 5 is load in tons, by $\sqrt{5} = 2.23''$, call it $2\frac{1}{4}''$. Hole through trunion for hook 2". Bending moment at center of trunion will be $\frac{10,000 \times 5}{2 \times 2} = 12,500$ in. lbs. Section modulus for section assumed is $\frac{2.5 \times (2.5)^2}{6} = 2.6$. Actual fibre stress is $\frac{12,500}{2.6} = 4,800\#$. This is satisfactory. Bending moment on trunion bearing will be $5,000 \times 0.25 = 1,250$ in. lbs. $\frac{1,250}{12,000} = 0.104$. From tables d is $1''$. Area of bronze washer for hook is $\frac{\pi((4)^2 - (2)^2)}{4} = 9.42$ sq. in. Pressure per sq. in. of area will be $\frac{10,000}{9.42} = 1,060\#$. This is satisfactory for steel on bronze for reverse and intermittent pressures, according to Hütte.

33.-Sheave Pins and Bearing.- Bending moment on pin in floating sheave will be $\frac{10,000 \times 5}{2 \times 2} = 12,500$ in. lbs. $\frac{12,500}{12,000} = 1.05$. From tables d should be $2\frac{1}{4}''$. Unit pressure per sq. in. of projected area of bearing is $\frac{10,000}{2.25 \times 4.5} = 1,000\#$. This is satisfactory. Thickness of bronze bushing $\frac{1}{4}''$. Resultant force on the pin in fixed sheave is shown by Fig. 6, page 8 to be $EC = 10,000\#$. Bending moment will be $\frac{10,000 \times 8}{2 \times 2} = 20,000$ in. lbs. $\frac{20,000}{12,000} = 1.66$. From tables d is $2\frac{1}{2}''$. Unit pressure per sq. in. of projected area of bearing is $\frac{10,000}{2.5 \times 5} = 800\#$. Thickness of bronze bushing $\frac{1}{4}''$.

34.-Dispatch Lowering Brake.- A brake of the Weston type was selected.

α = angle of helix or screw thread $= 4° - 5'$.

h = pitch of screw $= \frac{1}{2}''$.

r = pitch radius of screw $= 1\frac{1}{8}''$.

$\tan\alpha = \frac{0.5}{2\times\pi\times 1.12} = 0.07.$

F = force exerted on hand wheel.

a = radius of hand wheel in inches $= 8\frac{1}{2}''$.

ρ = angle of friction $= \tan^{-1} 0.06 = 3^\circ - 26'$.

μ_1 = coefficient of friction between nut and disks $= 0.06$.

r_1 = mean radius of nut $= 1\frac{7}{8}''$.

K = axial thrust on screw and disks.

M = twisting moment on pinion $= 7{,}350$ in. lbs.

n = number of friction disks in sliding contact $= 15$.

R = mean radius of friction disks $= 3\frac{1}{4}''$.

μ_2 = coefficient of friction between disks $= 0.10$.

$\alpha+\rho = 7^\circ - 31'$. $\tan(\alpha+\rho) = 0.1316$.

$F\times a = K\cdot r\cdot\tan(\alpha+\rho) + \mu_1\cdot K\cdot r_1$.

$M = n\cdot\mu_2\cdot K\cdot R$. $K = \frac{7{,}350}{15\times 3.25\times 0.10} = 1{,}510\#$.

$F = \frac{(1{,}510\times 1.125\times 0.1316) + (0.06\times 1510\times 1.87)}{8.5} = 46\#$. This is within the capacity of one man. Pressure per sq. in. of disks will be $\frac{1{,}510}{35.73} = 42\#$.

35.-Safety Brake.-

R_1 = mean radius of friction face to left of ratchet $= 3\frac{1}{8}''$.

μ_1 = coefficient of friction between faces to left of ratchet = 0.3. This value is high as hard fibre is used.

α = angle of helix $= 7^\circ - 20'$.

ρ = friction angle between helix faces $= 3^\circ - 26'$.

$\alpha+\rho = 10^\circ - 56'$. $\tan(\alpha+\rho) = 0.193$.

r = pitch radius of helix $= 1\frac{7}{8}''$.

M = twisting moment on shaft $= 7{,}350$ in. lbs.

$$M = K \cdot r \cdot \tan(\alpha + \rho) + \mu_1 K \cdot R_1 .$$

$$7{,}350 = (K \times 1.875 \times 0.193) + (0.3 \times K \times 3.125).$$

$$K = \frac{7{,}350}{0.362 + 0.937} = 5{,}650\#.$$

This is for raising the load by motor. When the load tends to run away when being lowered by the motor or tends to run down when unsupported by the motor the safety brake interposed between the helix and the ratchet wheel will be called into action. As the twisting moment and axial thrust is practically the same as for the dispatch brake, the same number of friction discs and same dimension as for that brake will be used.

36.-Ratchet Wheel.- Diameter of wheel assumed as 10", 16 teeth and $1\frac{1}{2}$" face, pitch of teeth will be

$$p_c = \sqrt{\frac{16 \times \pi \times 7{,}350}{1.5 \times 3{,}000 \times 16}} = 2.26\text{"}, \text{ call it } 2\tfrac{1}{4}\text{"} .$$

Actual pitch $\frac{\pi \times 10}{16} = 1.98$", This will do as the value of S has been taken low. Thrust on pawl is $\frac{7{,}350}{5} = 1{,}470\#$. Bending moment on pawl rod is $\frac{1{,}470 \times 18.5 \times 10.5}{29} = 10{,}000$ in. lbs. $\frac{10{,}000}{15{,}000} = 0.66$. From tables $d = 1\frac{7}{8}$".

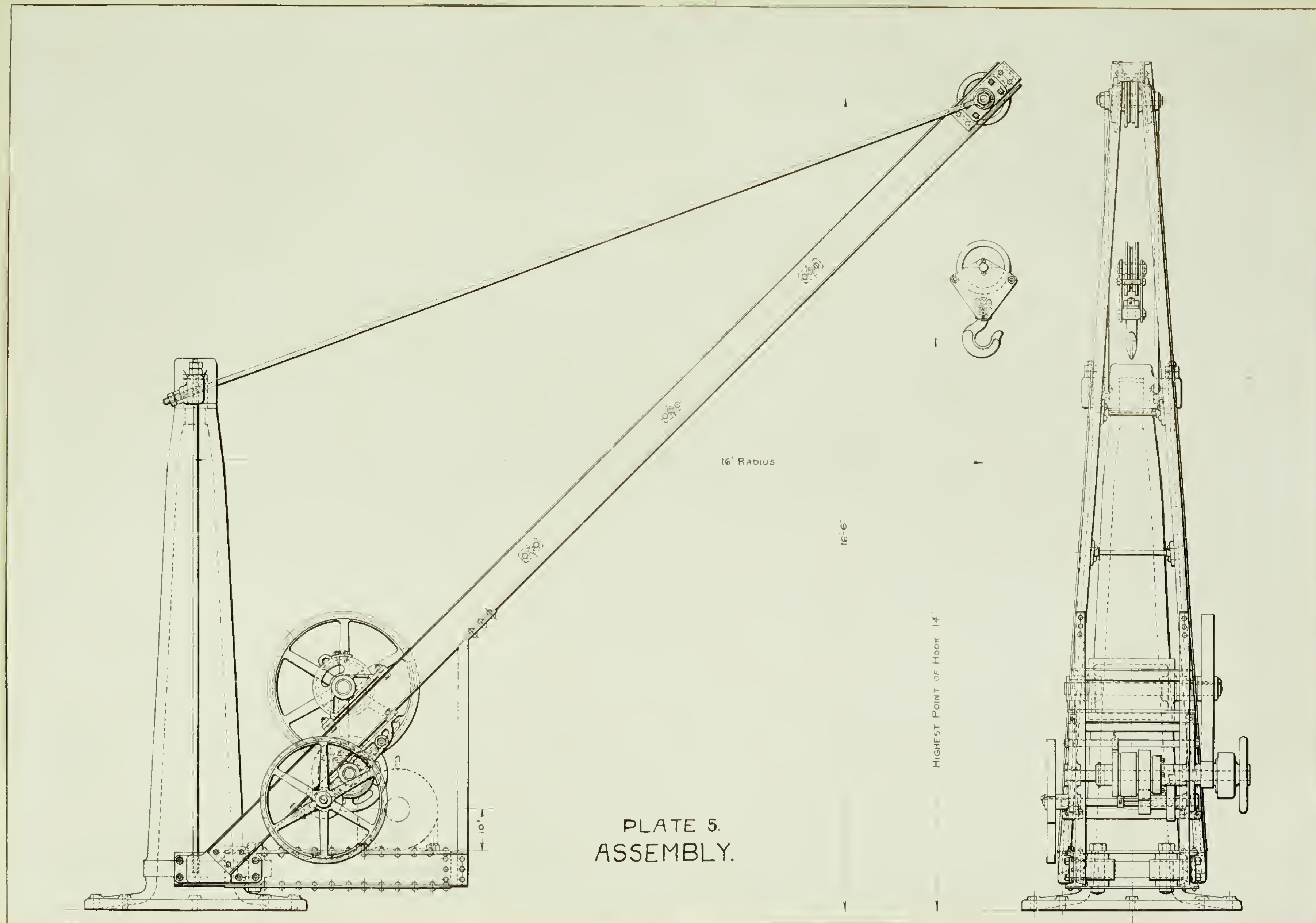
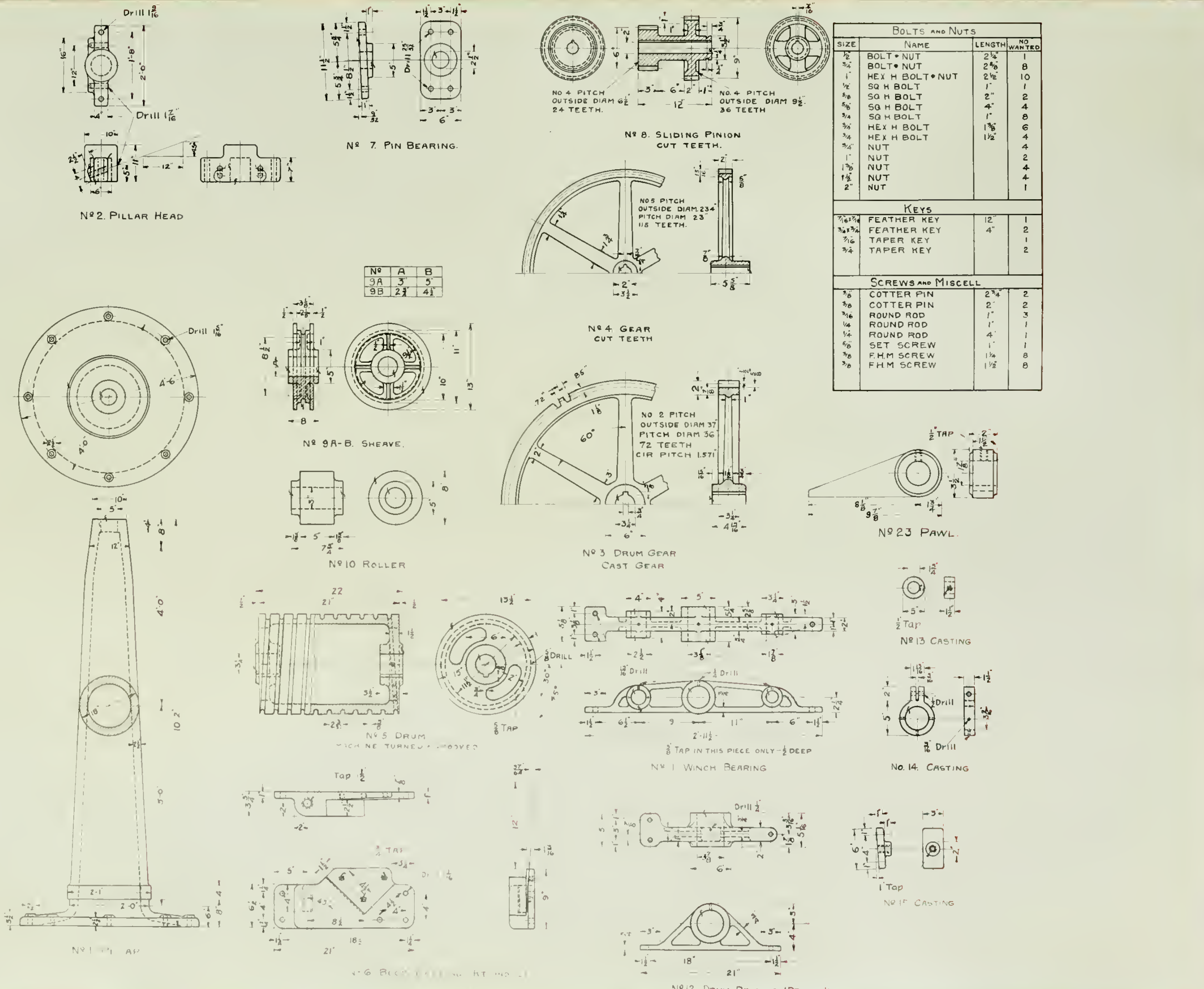


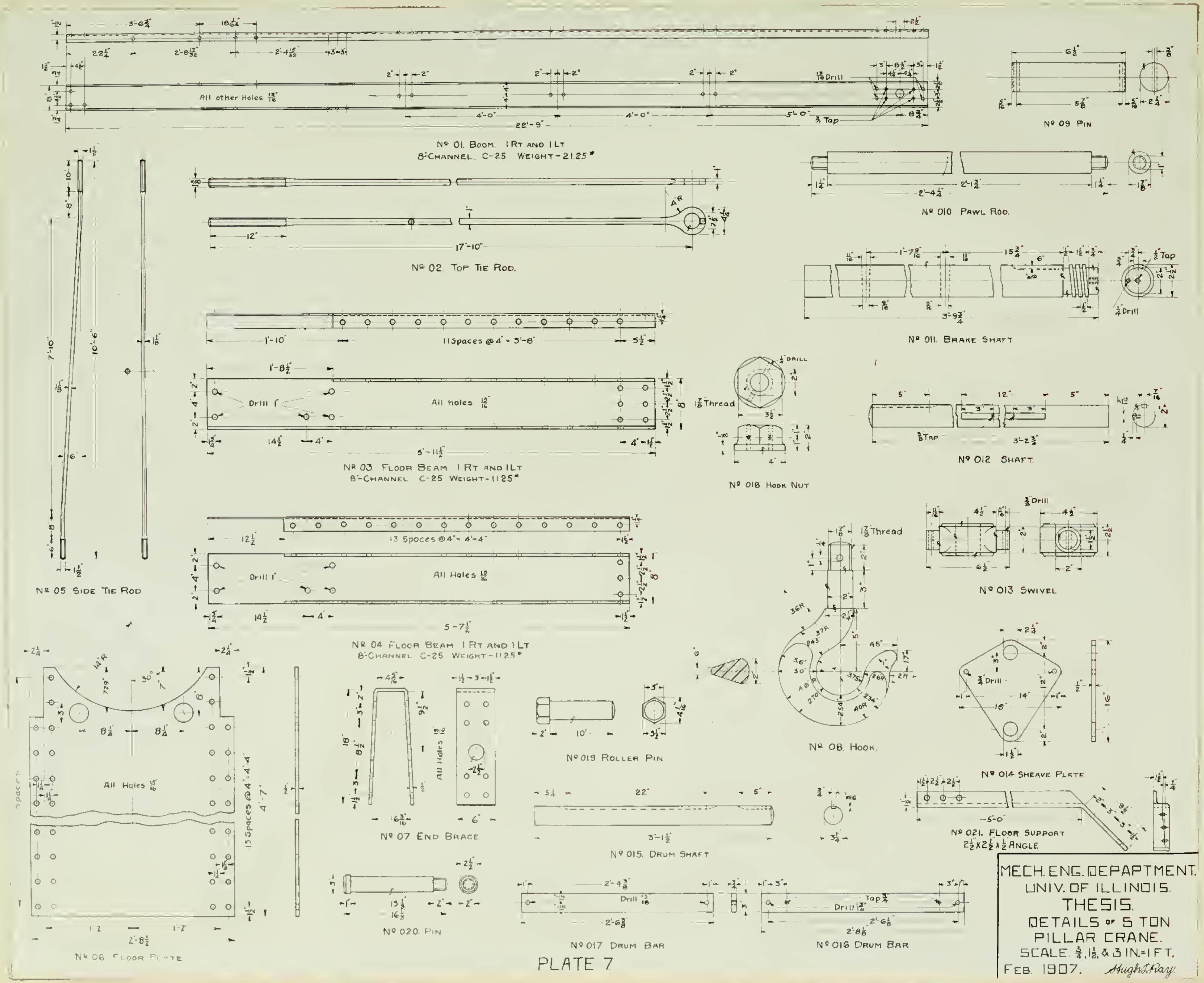
PLATE 5.
ASSEMBLY.



Nº	A	B
9A	3"	5"
9B	2¾	4½

Size	Name	Length	No Wanted
Bolts and Nuts			
½	BOLT • NUT	2¼"	1
¾	BOLT • NUT	2⅝"	8
1"	HEX H BOLT • NUT	2½"	10
½	SQ H BOLT	1"	1
⅝	SQ H BOLT	2"	2
⅝	SQ H BOLT	4"	4
¾	SQ H BOLT	1"	8
¾	HEX H BOLT	1⅜"	6
¾	HEX H BOLT	1½"	4
¾"	NUT		4
1"	NUT		2
1⅜	NUT		4
1½	NUT		4
2"	NUT		1
Keys			
7/16 x 7/16	FEATHER KEY	12"	1
¾ x ¾	FEATHER KEY	4"	2
7/16	TAPER KEY		1
¾	TAPER KEY		2
Screws and Miscell			
⅜	COTTER PIN	2¾"	2
⅜	COTTER PIN	2"	2
3/16	ROUND ROD	1"	3
¼	ROUND ROD	1"	1
¼	ROUND ROD	4'	1
⅝	SET SCREW	1"	1
⅜	F.H.M SCREW	1¼	8
⅜	F.H.M SCREW	1½"	8

PLATE 6



Nº 01. Boom. 1 Rt and 1 Lt
8"-Channel. C-25 Weight - 21.25#
Nº 09 Pin
Nº 010 Pawl Rod.
Nº 02. Top Tie Rod.
Nº 011. Brake Shaft
Nº 03. Floor Beam 1 Rt and 1 Lt
8"-Channel C-25 Weight - 11.25#
Nº 018 Hook Nut
Nº 012 Shaft.
Nº 05 Side Tie Rod
Nº 04 Floor Beam 1 Rt and 1 Lt
8"-Channel C-25 Weight - 11.25#
Nº 013 Swivel
Nº 08. Hook.
Nº 014 Sheave Plate
Nº 019 Roller Pin
Nº 07 End Brace
Nº 015. Drum Shaft
Nº 021. Floor Support
$2\frac{1}{2}$x$2\frac{1}{2}$x$\frac{1}{2}$ Angle
Nº 06 Floor Plate
Nº 020. Pin
Nº 017 Drum Bar
Nº 016 Drum Bar
PLATE 7
MECH. ENG. DEPARTMENT.
UNIV. OF ILLINOIS.
THESIS.
DETAILS of 5 TON
PILLAR CRANE.
SCALE. $\frac{3}{4}$, $1\frac{1}{2}$, & 3 IN. = 1 FT.
FEB. 1907.
Hugh T. Ray

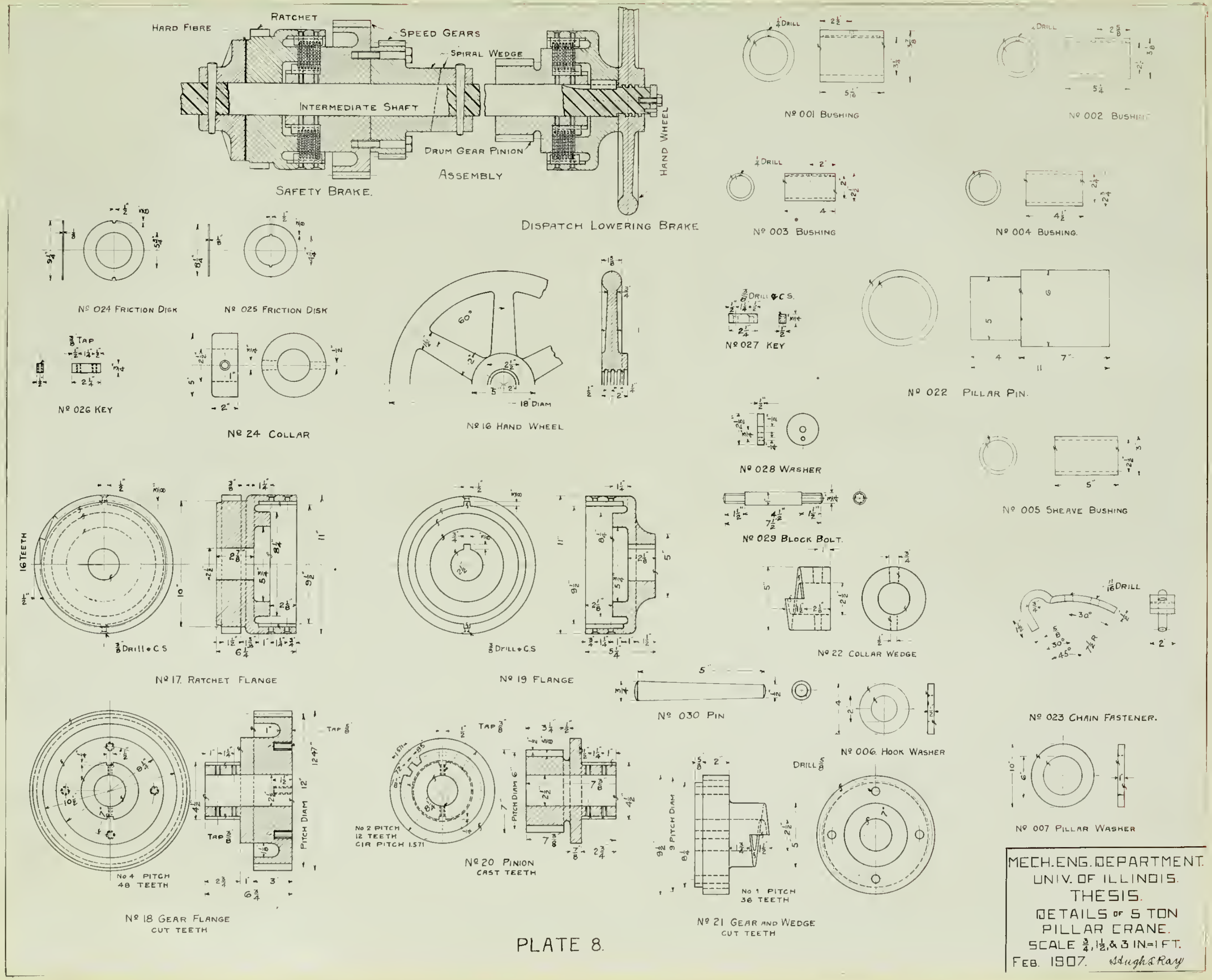


PLATE 8.

MECH. ENG. DEPARTMENT.
UNIV. OF ILLINOIS.
THESIS.
DETAILS OF 5 TON
PILLAR CRANE.
SCALE $\frac{3}{4}$, $1\frac{1}{2}$, & 3 IN = 1 FT.
FEB. 1907. Hugh & Ray

CHAPTER III.

Stock Lists.

-Cast Iron-

Piece Number	Name	Number Wanted	Plate Number
1	Pillar	1	6
2	Pillar Head	1	6
3	Drum Gear	1	6
4	Gear	1	6
5	Drum	1	6
6 Rt.	Boom Casting	1	6
6 Lt.	Boom Casting	1	6
7	Pin Bearing	2	6
8	Sliding Pinion	1	6
9-A	Sheave	1	6
9-B	Sheave	1	6
10	Roller	2	6
11	Winch Bearing	2	6
12 Rt.	Drum Bearing	1	6
12 Lt.	Drum Bearing	1	6
13	Collar	1	6
14	Collar	1	6
15	Casting	6	6
16	Hand Wheel	1	8
17	Ratchet Flange	1	8
18	Gear Flange	1	8
19	Flange	1	8
20	Pinion	1	8
21	Gear	1	8
22	Wedge Collar	1	8
23	Pawl	1	6
24	Collar	1	8

-Steel-

Piece Number	Name	Number Wanted	Plate Number
01 Rt.	Boom	1	7
01 Lt.	Boom	1	7
02	Top Tie Rod	2	7
03 Rt.	Floor Beam	1	7
03 Lt.	Floor Beam	1	7
04 Rt.	Floor Beam	1	7
04 Lt.	Floor Beam	1	7
05	Side Tie Rod	2	7
06	Floor Plate	2	7
07	End Brace	1	7
08	Crane Hook	1	7
09	Pin	1	7
010	Pawl Rod	1	7
011	Brake Shaft	1	7
012	Shaft	1	7
013	Swivel	1	7
014	Sheave Plate	2	7
015	Drum Shaft	1	7
016	Drum Bar	1	7
017	Drum Bar	1	7
018	Hook Nut	1	7
019	Roller Pin	2	7
020	Pin	1	7
021 Rt.	Floor Support	1	7
021 Lt.	Floor Support	1	7
022	Pillar Pin	1	8
023	Chain Fastener	1	8
024	Friction Disk	14	8
025	Friction Disk	14	8
026	Key	4	8
027	Key	4	8
028	Washer	1	8
029	Block Bolt	2	8
030	Pin	2	8

-Bronze-

Piece Number	Name	Number Wanted	Plate Number
001	Bushing	2	8
002	Bushing	2	8
003	Bushing	2	8
004	Bushing	1	8
005	Bushing	1	8
006	Hook Washer	1	8
007	Pillar Washer	1	8

-Rivets-

Size	Name	Length	Number Wanted
$\frac{3}{4}$"	R. H. Rivet	$1\frac{7}{8}$"	6
$\frac{3}{4}$"	R. H. Rivet	$2\frac{1}{8}$"	6
$\frac{3}{4}$"	R. H. Rivet	$2\frac{1}{4}$"	110
$\frac{3}{4}$"	R. H. Rivet	$2\frac{3}{8}$"	8
$\frac{3}{4}$"	R. H. Rivet	3"	12